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TITLE FOUR-WHEEL-DRIVE POWERTRAIN MODELS FOR REAL-TIME

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Four-wheel-drive Powertrain Models for Real-time Simulation

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Abstract

Dynamic vehicle powertrain models have been formulated by many researchers for particular applications, mainly the analysis of powertrain characteristics or vibration. In this paper, we formulate models of four-wheel-drive vehicle powertrains for the purpose of real-time simulation. While not encompassing the complete characteristics of powertrain operation, these models are applicable for use in vehicle simulators, and include both the differential and lock-up modes of four-wheel-drive transfer case operation.

The models of four-wheel-drive powertrains are applied to a particular vehicle, the US Army High Mobility Multipurpose Wheeled Vehicle or HMMWV, in order to compare and contrast both the simulated vehicle performance and the computational complexity of the models. The HMMWV powertrain incorporates a wide variety of components including: automatic transmission, torque converter, worm-gear differentials, transfer case for both differential and lock-up operation, and final-drive speed reducers at the wheel hubs. Thus the simulation of this vehicle indicates the wide application of the developed models.

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1 Introduction

Dynamic vehicle powertrain models have been formulated by a number of researchers for a variety of purposes ranging from investigations of the dynamics and vibrational characteristcs of the powertrain to studies of powertrain control; see for example Kotwicki (1982); Cho and Hedrick (1988); or Karmel (1986). The models developed herein are formulated for use in real-time vehicle dynamic simulators. The primary purposes of such simulators are to help understand operator-machine interaction as it applies to vehicles. In particular, aspects such as age, fatigue, narcotics, alcohol, disabilities, accident prevention training, vehicle and highway design, traffic control devices and transportation system automation are of considerable interest; see Stoner, et al. (1990). One important aspect is that it is desirable to minimize the required amount of component testing yet still adequately simulate vehicle behavior for the model purposes. As such, several assumptions made in deriving these models include

- The torque converter is always operating at its designed capacity.
- The torque converter and transmission gearing are considered inertialess.
- The wheels always remain in contact with the ground.
- The torque ratio of the *Torsen* differentials can be approximated by the function represented in Fig. 1.

We feel these assumptions are justifiable, since previous studies on torque converter dynamics; see Ishihara (1966), have shown the quasi-steady-state model to be adequate for engine velocity disturbances with a frequency content one-half the rotational frequency of the engine. Additionally, the assumption of an inertialess transmission should cause negligible errors in the predicted performance since the total inertia is small compared to that of the effective inertia of the vehicle.

2 Vehicle Model

While more complex vehicle models are employed in the vehicle simulators, in order to develop and exercise the powertrain models we have employed a vehicle model incorporating ten degrees of-freedom; including longitudinal and lateral displacement, roll, pitch and yaw rotations, flexible steering system dynamics, and tire rotational dynamics. The tire force model used in this study is a modified form of the Dugoff model; see Dugoff, et al. (1970) and Guntar and Sankar (1980). The Dugoff model calculates tire tractive forces as functions of the tire load, slip angle and slip ratio, thus accounting for the trade-off between longitudinal (braking/tractive) force and cornering force and reasonably accounts for tire behavior near the limits of adhesion.

3 Powertrain Models

Two types of four-wheel-drive powertrain models are derived in this study. The first is a powertrain incorporating a differential transfer case, and the second is a powertrain with a "locked-up" or rigid transfer case. With the exception of a Lagrange multiplier in the second model, which is used to enforce the rigid transfer case constraint, the two powertrain models are equivalent. The particular powertrain used for example is the US Army High Mobility Multipurpose Wheeled Vehicle, or HMMWV, a full-time four-wheel-drive vehicle which incorporates a lockable differential transfer case.

The formulation treats the powertrain as five lumped inertias, one representing the engine and the others representing the rotational inertias of the wheels. The lumped engine inertia includes the inertias of the crankshaft, torque converter impeller and the average inertia of the reciprocating pistons, while the lumped wheel inertias include the inertias of the transfer case, driveshafts, differential gearing, axles, disk brakes, final-drive gearing, wheel hubs and tires. Thus, expressing the second-order dynamics as

$$[\mathbf{J}]\left\{\ddot{\theta}\right\} = \{F\} + [\phi_\theta]^T\{\lambda\} \tag{1}$$

where, by assuming rigid connections between lumped inertias, the condensed form of the inertia matrix can be written as

$$[\mathbf{J}] = \begin{bmatrix} J_E & 0 & 0 & 0 & 0 \\ 0 & J^*_{WLF} & \frac{1}{8}N^2_{FD}N^2_D J^*_{FS} & \frac{1}{32}N^2_{FD}N^2_D J_T & \frac{1}{32}N^2_{FD}N^2_D J_T \\ 0 & \frac{1}{8}N^2_{FD}N^2_D J^*_{FS} & J^*_{WRF} & \frac{1}{32}N^2_{FD}N^2_D J_T & \frac{1}{32}N^2_{FD}N^2_D J_T \\ 0 & \frac{1}{32}N^2_{FD}N^2_D J_T & \frac{1}{32}N^2_{FD}N^2_D J_T & J^*_{WLR} & \frac{1}{8}N^2_{FD}N^2_D J^*_{RS} \\ 0 & \frac{1}{32}N^2_{FD}N^2_D J_T & \frac{1}{32}N^2_{FD}N^2_D J_T & \frac{1}{8}N^2_{FD}N^2_D J^*_{RS} & J^*_{WRR} \end{bmatrix}.$$

In this equation, J_T is the inertia of the transmission, and the terms J^*_{FS}, J^*_{RS}, J^*_{WLF}, J^*_{WRF}, J^*_{WLR}, and J^*_{WRR} are defined as

$$J^*_{iS} = \frac{1}{4}J_T + J_{iS} \qquad i = F, R \tag{2}$$

and

$$J^*_{Wij} = \frac{1}{16}N^2_{FD}N^2_D J_T + \frac{1}{4}N^2_{FD}N^2_D J_{jS} + N^2_{FD}J_{ij} + J_{Wij} \qquad i = L, R, j = F, R \tag{3}$$

where J_T is the lumped inertia representing the transmission and transfer case, J_{FS} and J_{RS} are the inertias of the front and rear driveshafts, respectively, J_{LF}, J_{RF}, J_{LR} and J_{RR} are the lumped inertias of the brake disks and axles, and J_{WLF}, J_{WRF}, J_{WLR} and J_{WRR} are the inertias of the combined tire, wheel hub and final-drive assembly.

3.1 Engine Model

The powertrain model incorporates a very simplified model of a diesel engine. A bilinear surface fit of the engine's steady-state performance map is used to approximate engine torque as a function of engine speed and fuel consumption. While noise and vibration emminate from the engine, the accurate simulation of these phenomena are not essential to driver behavior. It is significantly easier to add such sound and vibration later in an ad hoc manner as opposed to the use of more complex engine models which would require significantly more model development and verification testing. In this particular implementation, the calculated engine torque is a steady-state approximation to the actual torque as a function of engine speed, $\dot{\theta}_E$, and the governed fuel rack position, ϕ_f. This is expressable as

$$T_E = \mathcal{T}_E\left(\dot{\theta}_E, \phi_f\right). \tag{4}$$

A simplified dynamic model of the diesel engine can be written

$$J_E \ddot{\theta}_E = T_E - T_{ACC} - T_{TCi} \tag{5}$$

where T_{ACC} are the torque losses due to engine accessories, and T_{TCi} is the absorbed input torque of the torque converter. In this study, the engine accessory losses are modeled as an algebraic function of the engine speed.

3.2 Torque Converter Model

The torque converter model employed in this study is based on the cubic spline curvefits of the torque ratio vs. speed ratio and the torque capacity factor vs. speed ratio curves. These curves are standard methods of representing the steady-state characteristics of torque converters. An example curve is diagrammed in Fig. 2. The torque capacity factor, K, represents the relationship of input torque and speed for a converter of a particular size and internal blade geometry. The absorbed input torque, T_{TCi}, at steady-state conditions is equal to the square of the inverse capacity factor multiplied by the square of the input speed, or

$$T_{TCi} = \frac{1}{K^2} \dot{\theta}_E^2. \tag{6}$$

The output torque can be found from the input torque by multiplying by the torque ratio, γ, as follows

$$T_{TCo} = \gamma T_{TCi}. \tag{7}$$

3.3 Transmission Model

The transmission is modeled as single input and output shafts which are connected thru a set of gear reducers by ideal clutches. The torque converter output is linked to the transmission input, and a single inertia, J_T, is concentrated at the output shaft. The speed relationship between the torque converter output angular velocity, $\dot{\theta}_{TCo}$, and the transmission output angular velocity, $\dot{\theta}_T$, is given by

$$\dot{\theta}_{TCo} = N_j \dot{\theta}_T \tag{8}$$

where N_j is the speed ratio of the j^{th} gearset

The torque converter and transmission output torques are related by the following expression

$$T_T \;=\; \eta_j N_j T_{TC_o} \tag{9}$$

where T_{TC_o} and T_T are the torque outputs, and η_j is the efficiency of the j^{th} gearset.

3.4 Transfer Case

Two different transfer case models are employed in the current study; the first being a differential transfer case, and the second being a "locked-up" or rigid transfer case. For both transfer case types, the angular velocity of the transmission shaft is related to the angular velocities of the driveshafts by the equation

$$\dot{\theta}_T \;=\; \frac{1}{2}\left(\dot{\theta}_{FS} + \dot{\theta}_{RS}\right) \tag{10}$$

where $\dot{\theta}_{FS}$ and $\dot{\theta}_{RS}$ represent the front and rear driveshaft angular velocities, respectively. For the rigid transfer case, there is the additional constraint equation

$$\dot{\theta}_{FS} \;=\; \dot{\theta}_{RS} \tag{11}$$

which expresses that the front and rear driveshafts must rotate with the same speed when the transfer case is locked.

3.5 Differential Model

The vehicle modeled in this study uses worm-gear, or *Torsen*, differentials. The Torsen differential model differs from the conventional differential model since it allows a variable torque-split based on the difference of the output shaft speeds. In the implementation applied in the current study, the magnitude of the torque split is found from the curve diagrammed in Fig. 1. The torque split can also be modeled using an Euler-Lagrange derivation in combination with a friction model [Freeman and Velinsky, 1989], however, application of this more complicated model is of questionable value to this implementation.

Like the transfer case, the input shaft velocity of the differential can be related to the axle velocities by the following equation,

$$\dot{\theta}_{iS} \;=\; \frac{1}{2} N_D \left(\dot{\theta}_{Li} + \dot{\theta}_{Ri}\right) \qquad i = F, R \tag{12}$$

where $\dot{\theta}_{iS}$ signifies either the front or rear driveshaft velocity, and $\dot{\theta}_{Li}$ and $\dot{\theta}_{Ri}$ are the corresponding left and right axle velocities. The curve shown in Fig. 1 provides for a 2:1 torque bias ratio between the two axles, with the slower axle receiving the larger amount of torque. Provided that α represented the torque-split found from this curve, the torque output to the slower axle is expressable as

$$T_{slow} = \alpha \eta_D N_D T_{iS} \tag{13}$$

and the torque for the faster axle as

$$T_{fast} = (1-\alpha)\eta_D N_D T_{iS} \tag{14}$$

where T_{iS} is either the front or rear driveshaft torque, N_D is the speed ratio and η_D is the efficiency of the differential.

3.6 Final Drive Gearbox Model

The vehicle used for modeling purposes has the somewhat unusual arrangement of having final-drive gearboxes mounted at the ends of the suspension arms, for the purpose of increasing overall ground-clearance. The angular velocity relationship of the wheel and the axle is simply given by

$$\dot{\theta}_{ji} = N_{FD}\dot{\theta}_{Wji} \tag{15}$$

where $\dot{\theta}_{ji}$ is the angular velocity of the connected axle, $\dot{\theta}_{Wji}$ is the velocity of the tire, and N_{FD} is the final-drive speed ratio. In a similar manner, the torque at the wheel can be expressed as a simple ratio of the axle torque, T_{ji}, by the following equation

$$T_{Wji} = \eta_{FD} N_{FD} T_{ji} \tag{16}$$

where η_{FD} is the efficiency of the final-drive gearing and T_{Wji} is the wheel torque.

4 Testing Results

Two tests were selected to verify performance of the models by comparison with non dynamic data. The first test being a comparison of vehicle straightline tractive performance throughout the operational range of the vehicle, the second being a test of tractive performance during low-speed, tight cornering. These tests were analysed by Koga (1988) as a means of comparing different type of four-wheel-drive powertrains and their basic characteristics of operation.

The straightline tractive tests were performed with the transfer case in the unlocked, or differential, position and in the high-locked, or rigid, position. A restrictive force was applied to the vehicle at the trailer hitch. The test results are shown in Fig. 3. As expected, the differential transfer case model exhibits equal torque proportioning between the front and rear axles. While the locked transfer case model displays a greater proportion of torque being transmitted to the front axle at low to moderate tractive loads, while at higher tractive loading the torque proportioning becomes greater at the rear axle. These results are similar to those of Koga (1988), however that study shcwed much more dramatic changes in the torque proportioning of the locked transfer case. The change in front/rear torque proportioning is due to changes in wheel loading, since the applied tractive load creates a moment about the pitching axis of the vehicle. Without the effects of weight transfer, the tractive torque proportioning follows the static wheel load proportions. In the study of Koga (1988), static wheel loading was proportionally much higher on the front wheels, than it is for the HMMWV modeled in this study.

The low-speed, cornering tests were run using the same conditions as the straightline tractive tests except that a steering input of 20° at the front tires was used. For the HMMWV, this results in a turning radius of approximately 48 feet. Results at low tractive torques follow the results of Koga (1988), however as the total tractive torque increases to a value that causes the inside rear tire to skid, the front/rear torque proportioning returns to the values shown by the straightline data.

Computation times for the model are given in Table 1. The integration algorithm used is based on LSODE, a variable-order, variable-stepsize Euler backward difference technique. Despite the fact that the algorithm is not optimized for speed, the model of the vehicle and powertrain incorporating either the lock-up or differential transfer case is computable in real-time on a 25 MHz 80486 PC.

Differential Model	10.44 sec.
Lock-up Model	11.04 sec.

Table 1 Time to compute 15 seconds of simulation output.

5 Conclusions

Two models of four-wheel-drive powertrains were formulated in this study for application in a vehicle simulator. The models were formulated to simulate the lock-up and differential modes of transfer case operation for a four-wheel-drive vehicle. Other types of four-wheel-drive vehicle transfer cases could be readily incorporated since the two modes modeled represent the extremes of normal transfer case operation. While the models presented do not encompass the complete characteristics of vehicle powertrain operation, and as of yet have not be compared to actual vehicle data, they do incorporate an adequate level of behavioral characteristics for use in a vehicle simulator.

We have found that the characteristics of the differentials is dependent on the vehicle wheel loading, and that weight transfer effects change the relative values of tractive torque proportioning. Finally, tt is important that the models can be computed in real-time, in order to respond to operator inputs, we have shown this to be achievable with current personal computer hardware.

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